J. Mesin Vol. 29, No. 2, 2023, 146-162

Multi-Objective Finned-Tube Heat Exchanger Optimization Using a Genetic Algorithm

Nadiyah Rizki Suyatna[1], Firman Bagja Juangsa[1*] & Prihadi Setyo Darmanto[1]

[1]Fakultas Teknik Mesin dan Dirgantara
Institut Teknologi Bandung, Jalan Ganeca 10, Bandung, Indonesia
[*]Email: firman.juangsa@itb.ac.id

Abstract. Heat exchangers are a significant component in many industries, particularly in energy conversion systems. The design of heat exchangers itself is a complex process because it involves experience-based decisions, numerous variables and parameters, and some of them are competing with each other. Genetic Algorithms (GAs) are one of the first evolutionary algorithms which remains one of the most extensively used non-linear optimization methods today. This study explores the implementation of Non-Dominated Sorting Genetic Algorithm II (NSGA-II) for thermal design and optimization of a finned-tube heat exchanger. The chosen objective functions were minimizing the heat exchanger volume and minimizing the air side pressure drop. The decision variables for the design were tube outer diameter, number of tube rows, fin pitch, unit height, and unit width. The calculated parameters and estimated cost of both preliminary design and optimized design were also compared. The optimized design offered a bigger alternative design while meeting all the constraints according to standards and industrial needs. The annualized cost of the optimized design is only 30.4% of the preliminary design, and the air pressure drop can be reduced to 19.5% of the preliminary design, with a 12.4% increase in volume.

Key words: *cooling coil; finned-tube heat exchanger; genetic algorithm; multi-objective optimization; thermal design*

1 INTRODUCTION

It is known that gas turbines may lose a significant amount of their power output when the ambient air increases and this is a significant drawback for tropical climates where the average annual temperature ranges from 25–35°C like Indonesia. When the ambient temperature increases, the density of the intake air decreases, resulting in a lesser mass flow rate of air, which can decrease the turbine power output [1]. Compressor efficiency is also affected by the ambient air temperature. When the ambient temperature increases, the power demand for the inlet air compression also increases. The turbine's performance will drop rapidly because of the decreasing net electric power. For a gas turbine OPRA OP-16, for example, may lose about 9% of its nominal

Diterima 27 November 2023, Direvisi 22 Desember 2023, Diterima untuk dipublikasikan 29 Desember 2023

ISSN:0852-6095, DOI: 10.5614/MESIN.2023.29.2.4

power output when the ambient temperature increases from 15°C to 25°C and could reach up to 18% loss when the ambient temperature reaches 35°C [2].

One way to address this problem is by reducing the inlet air temperature using a heat exchanger with chilled water as the cooling media. The chilled water may be produced using a separate chiller during low-cost times and stored in a thermally isolated storage or using chilled water from the already available water cooling or building air conditioning systems. However, heat exchanger design is a complicated procedure that includes geometrical characteristics and operating standards, cost calculation, and optimization. The design of heat exchangers is a complex process, not only because of the mathematical calculations but also because the process requires experience-based decisions in various stages and numerous qualitative judgments [1]. The qualitative judgments might be based on past design experience, manufacturing capability, and industrial requirements. Some researchers used algorithms to optimize a heat exchanger during the design stage. Extensive advanced optimization techniques were applied. Trial and error were conducted in various parameter designs and operations, and it is very useful for the industrial applications [3].

The study of optimization using the particle swarm algorithm in a double pipe employed micro-finned tubes using a number of micro-fins from 10 to 60, micro-fin height varying up to 0.5 mm, and the micro-fin helix angle between 5 and 30° [4]. Grey wolf optimization algorithm could reduce total cost using relatively low computing time [5]. The elitist-Jaya algorithm in the first case was reduced by 32.86%, and in the second case reduced by 5.21% from the simulations result compared to original design for continuous parameter optimization [6]. H. Zareaa et. al proposed Multi-Objective optimization of Bees Algorithm Hybrid and Particle Swarm purposed to acquire the maximum effectiveness, and the minimum cost was simultaneously employed using seven decision parameters such as length in hot and cold side, frequency of fin, number of fin layers, thickness of fin, fin height, and fin lance length [7]. Gravitational search algorithm was developed from economic point of view. The algorithm was applied to two cases compared to the original data and other algorithms. The total cost could be reduced by 22.3% as compared to the original data The Gravitational search algorithm could be successfully applied for design optimization [8].

From several studies that have been conducted, optimization of the tube fins heat exchanger (HE) design is still widely accepted. Therefore the exploration of design optimization of HE tube fins will be studied in this study. The focus of this study is to explore the implementation of Non-Dominated Sorting Genetic Algorithm II (NSGA-II) for thermal design and optimization of a finned-tube heat exchanger. The chosen objective functions were minimizing the heat exchanger volume and minimizing the air side pressure drop. The decision variables for the design were tube outer diameter, number of tube rows, fin pitch, unit height, and unit width. The calculated parameters and estimated cost of both preliminary design and optimized design were also be compared.

2 MATERIALS AND METHOD

2.1. Materials

The case study for this research is to design a cooling coil for a gas turbine inlet air, assigned to the electric generation for offshore oil plant. The turbine used is OPRA OP-16 which nominally can generate 1883 kWe. It is a single-shaft all-radial gas turbine for industrial, commercial, marine, and oil & gas applications. Since its market introduction in 2005, over 140 generator sets based on the OP-16 gas turbine have been delivered worldwide. The OP-16 gas turbine features a single-stage centrifugal compressor with a pressure ratio of 6.7:1.

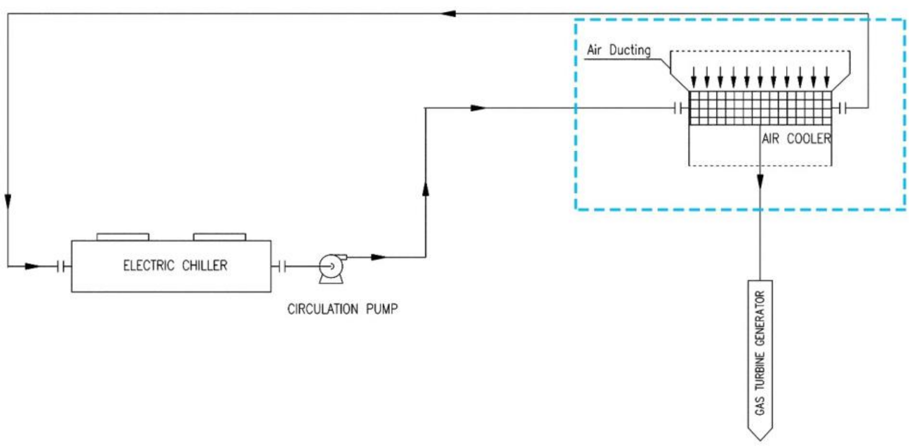

Figure 1 Schematic of gas turbin inlet air cooling system.

Meanwhile, the air cooling design is a fins tube heat exchanger (FTHE) type using chilled water produced by a chiller available at the plant for air conditioning needs as well. This FTHE is designed to reduce air temperature by 8.81 kg/s from 35°C to around 25°C with chilled water at 20°C as it enters the air cooler. With a chilled water flow rate of 5.04 kg/s, it is expected that the thermal duty of this air cooler is 88.74 kJ. In general, the technical specifications for the geometric parameter ranges of the designed air cooler are presented in Table 1, and these parameters will be optimized using Non-Dominated Sorting Genetic Algorithm II (NSGA-II) in order to match the design constraints mentioned in Table 2[9].

Table 1 Geometric ranges of the designed air cooler.

Parameters	*Values range*
Outer diameter (D_o)	0.0103 m to 0.0133 m
Number of row (N)	3 up to 8
Fin pitch per inch (f_p)	8 up to 14 fpi
Height of the cooler unit (H)	0.5 to 1.5 m
Width of the cooler unit (W)	0.75 to 3 m

The constraint values of air and water velocities and pressure drops are very important to determine in order to obtain the optimum air cooler geometry [10-13]. The ASHRAE 62.1 states that the standard maximum air pressure drop for finned-tube cooling coils is 0.75 inH_2O (around 187 Pa)[12,14].

Table 2 Design constraints.

Parameters	Value and unit
Maximum water flow velocity in copper tubes	1.8 m/s
Maximum air flow velocity across tubes	4 m/s
Maximum water side pressure drop	71.7 kPa
Maximum air side pressure drop	187 Pa

It should be noted that apart from the geometric parameters that must be optimized, there are two parameters that depend on other parameters, namely longitudinal and transverse tube pitch whose value depends on the value of the tube outer diameter. The material chosen for the pipe is copper, while aluminum is used for the fins, with a thickness of 0.12 mm.

2.2. Method

Optimization of the air cooler geometry is carried out using the multi-objective Genetic Algorithm method. The Genetic Algorithm (GA) is one of the first evolutionary algorithms. It draws inspiration from biological evolution based on Charles Darwin's theory of natural selection. A chromosome is a representation of the design point linked with an individual. After completing the genetic procedures of crossover and mutation, the chromosomes of the fitter individuals are passed on to the next generation at each generation. The fittest individual represents the optimal solution for the optimization problem[15]. A multi-objective problem has numerous objectives that must be optimized at the same time while being constrained by a number of inequality or equality constraints. In every iteration, individuals are evaluated based on their objective function. For example, in a minimization problem, the fittest individuals have the lowest numerical values of the objective functions. The fitness function is the ability of an individual to compete with other individuals and it is used to determine the individual's fitness level in the population. It is used to transform the numerical value of objective functions into a measure of relative fitness, thus:

$$F(x) = g(f(x)) \tag{1}$$

where f is the objective function, g transforms the value of the objective function to a non-negative number, and F is the resulting relative fitness.

The general form of a multi-objective optimization problem can be mathematically expressed in Equation (2) [15]:

$$\left.\begin{array}{lll} \text{Minimize or maximize} & f_m(x) & m = 1, 2, \ldots, M; \\ \text{Subject to} & g_j(x) \geq 0 & j = 1, 2, \ldots, J; \\ & h_k(x) = 0 & k = 1, 2, \ldots, K; \\ & x_i^{(L)} \leq x_i \leq x_i^{(U)}, & i = 1, 2, \ldots, n. \end{array}\right\} \tag{2}$$

where x is a vector of n design variables given by

$$x = \begin{bmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{bmatrix} \tag{3}$$

There are M objective functions given by

$$f(x) = \begin{bmatrix} f_1(x) \\ f_2(x) \\ \vdots \\ f_M(x) \end{bmatrix} \tag{4}$$

where each objective function can be either minimized or maximized. There are constraint functions $g_j(x)$ and $h_j(x)$ with J inequality and K equality constraints associated with the problem. The last set of constraints are called variable bounds, where the decision variable x_i needs to be between a lower bound $x_i^{(L)}$ and an upper bound $x_i^{(U)}$. Since multi-objective optimization requires the simultaneous optimization of two or more objective functions, it results in a set of optimal solutions known as the Pareto optimal solutions or Pareto front. In this study, multi-objective optimization of heat exchanger design using NSGA-II will be conducted with two objective functions (to minimize air pressure drop $f_1(x)$ and minimize heat exchanger volume $f_2(x)$) mathematically expressed by Equations (5) and (6).

$$f_1(x) = \frac{G_a^2}{2\rho_{a,i}}\left[\frac{A_o}{A_{airflow}}\frac{\rho_{a,m}}{\rho_{a,i}}f_a + (1+\sigma^2)\left(\frac{\rho_{a,i}}{\rho_{a,o}} - 1\right)\right] \tag{5}$$

$$f_2(x) = W\ L\ H \tag{6}$$

and the lower bound $x_i^{(L)}$ and upper bound $x_i^{(U)}$ for each variable is presented in Table 1, while the constraints parameters are laid out in Table 2. The optimization process using NSGA-II in MATLAB will be based on the mathematical modeling and the results of preliminary design by traditional approach and optimized design by genetic algorithm will be compared.

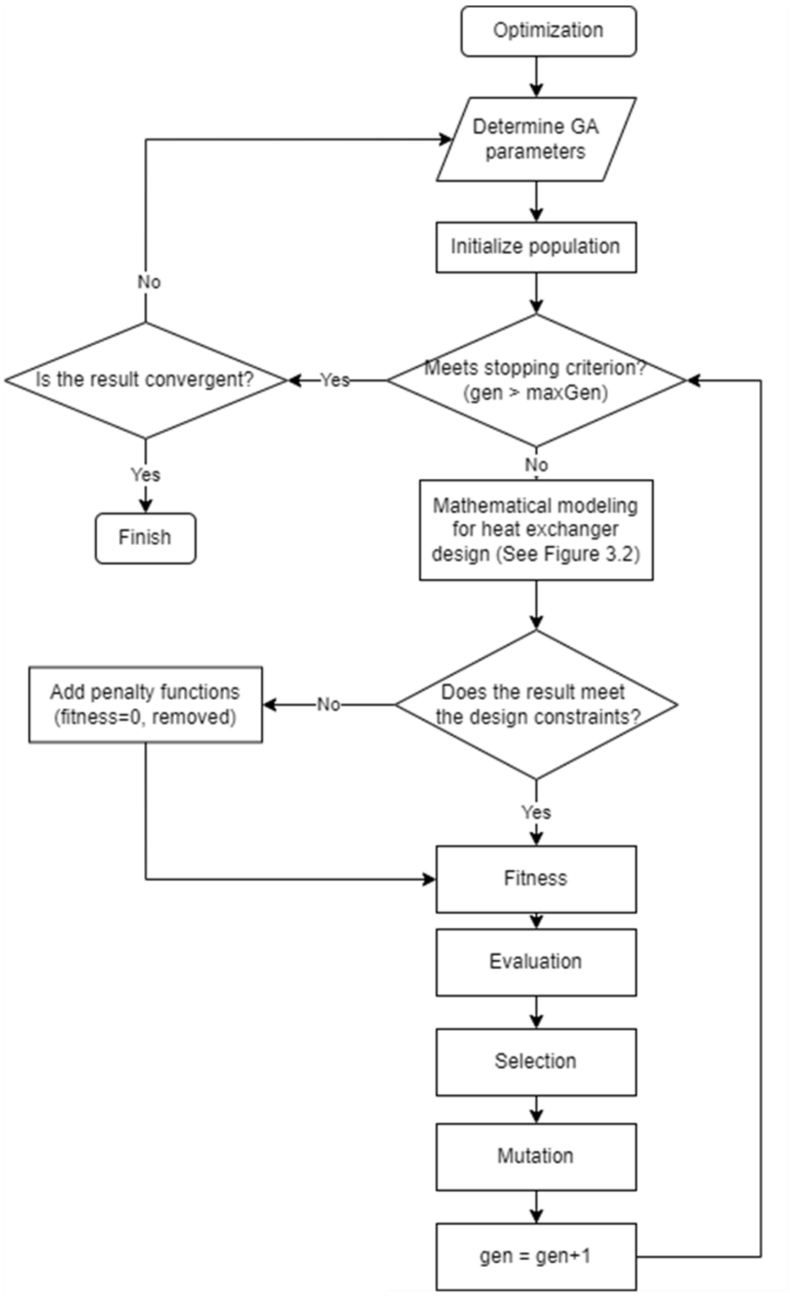

Figure 3 Multi-objective Optimization using NSGA-II in MATLAB.

The multi-objective genetic algorithm function *gamultiobj* in MATLAB uses a controlled elitist genetic algorithm which is a variant of NSGA-II [16]. NSGA-II itself is a modified version of non-dominated sorting genetic algorithm (NSGA) [11] which eliminated higher computational complexity, lack of elitism and the need for specifying the sharing parameter. The flow chart calculation of this multi-objective Optimization using NSGA-II in MATLAB is presented in Figure 3 [9].

3. AIR COOLER OPTIMIZATION DESIGN CALCULATION

3.1. Preliminary Design of Air Cooler Dimension

To be able to start the process of optimizing the air cooler geometry with the multi-objective genetic algorithms method, the geometry of the initial design results is needed. This preliminary design geometry is generated from calculations using traditional approach in HE design, namely the LMTD (logarithmic mean temperature difference) method [9]. Based on the design data previously provided as well as fluid properties presented in Table 3 and the parameter limits written in Table 1 and Table 2, the results of this traditional design are given in Table 4 with the thermal design parameters are mentioned in Table 5.

Table 3 Fluid properties used in the air cooler design.

Fluid properties	Water	Air
Volumetric flowrate [m^3/h]	18.20	27226.00
Mass flowrate [kg/s]	5.04	8.81
Density [kg/m^3]	997.00	1.18
Cp [J/kg.K]	4182.30	1007.22
Dynamic viscosity	0.90	18.71
Thermal conductivity [W/m.K]	0.61	0.03
Prandtl number	6.27	0.71
Inlet temperature [°C]	20.00	35.00
Outlet temperature [°C]		25.00
Fouling factor	0.0001	0.0004
Heat duty [J]	88743.95	88743.95

To ensure that the design results using the traditional approach are not fatally wrong, the calculated values are verified using the software built by Prof. Dr. Risto Ciconkov from the Faculty of Mechanical Engineering of St. Kiril and Metodij University, Macedonia [17]. The selection and rating of the heat exchanger works differently in the software, and the dimension W and H of the heat exchanger will be calculated automatically by the software instead of them being the input. Among the differences of the software evaluation results compared to the current method is that the software has heat load and number of tubes as input and heat exchanger dimension, air flowrate, and water flowrate as output. Therefore, the validation of the data is done by doing the heat exchanger selection using the software and inputting the data from the software to the current mathematical modeling and comparing the results.

Table 4 Preliminary design of air cooler geometries.

Parameters	Values and unit	
Tube outside diameter	13.3	mm
Tube rows	3	
Number of tubes	72	
Unit width	1.1	m
Unit length	0.09	m
Unit height	0.8	m
Fins pitch	2.116	mm
Numbers of fin per inch	12	fpi
Number of fins	521	

The heat exchanger characteristics for verification is similar to the preliminary design as shown in Table 4. The parameters are used as reference to be compared with the current mathematical modeling result as shown in Table 5. The percentage difference between each method shows that the current mathematical modeling is considered acceptable.

Table 5 Design results comparison.

Parameters	Software Ref. [17]	Calculation Results	Difference (%)
Heat transfer area [m^2]	53.78	53.81	-0.051
Overall heat transfer [W/m^2.K]	219.02	235.31	-7.43
Water velocity [m/s]	1.560	1.557	0.18
Air face velocity [m/s]	5.709	5.708	0.01
Air pressure drop [Pa]	164.60	155.67	5.43

3.2. Air Cooler Dimension Optimization Using NSGA-II

As mentioned previously, the output of the main algorithm include a set of plots as shown in Figure 4. Figure 4a shows the Pareto front which confirms the hypothesis that there is a trade-off or competition between the two objectives: heat exchanger size (volume) and air side pressure drop. Figure 4b shows the score histogram and the minimum and maximum value of each objective function. Figure 4c shows the average distance between individuals at each generation, which is a good measure of the diversity of a population. If the distance between individuals are too high, the result will be too diverse to produce an optimum solution. However, if the distance between individuals are too small, it will hinder the progress of lowering the fitness value. After several iterations of the GA parameter by changing the crossover probability, mutation result, and initial population size, the current set of parameters give the most satisfying result. Figure 4d shows that with each generation, the minimum value for the objective function converges. The graph stops at 100 generation because the number of maximum generation is set to be 100.

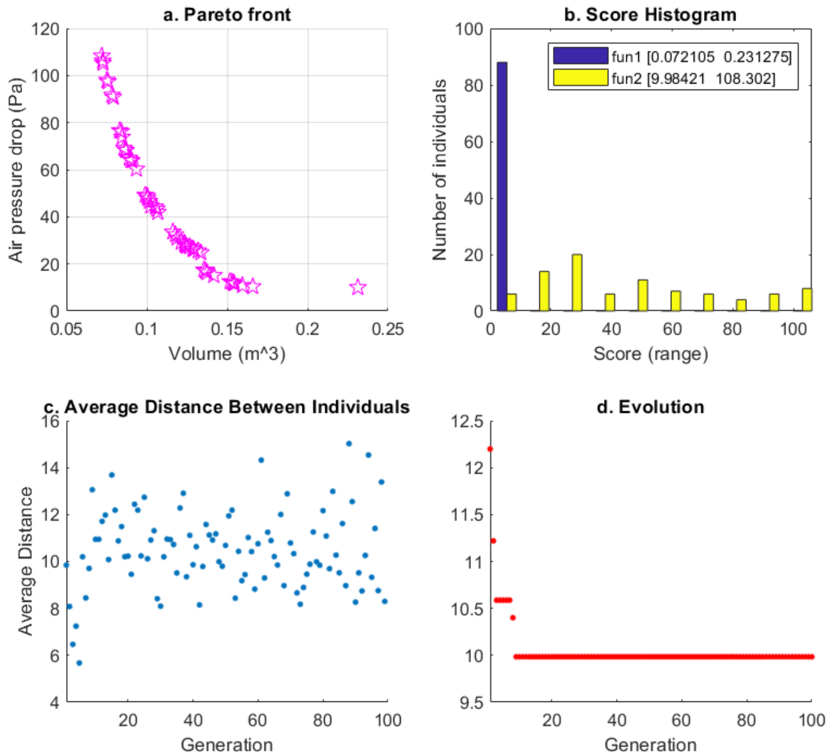

Figure 4 Multi objective optimization results.

All sets of Pareto optimal solutions are compiled and three optimal points will be chosen as shown in 5a. The influence of each decision variable on each objective will be observed using three cases of point A, B, and C. Point A represents a solution with the highest air pressure drop but smallest volume while point C represents a solution with the lowest air pressure drop but with the biggest volume. Point B lies in between two objectives as the middle point. The variation of two objectives from three optimal Pareto solutions are presented in Figure 5b to 5f. The influence of each parameter such as the tube outside diameter, tube rows, fin pitch as well as unit height and weight will be discussed. Furthermore the HE characteristics and calculated parameter values of the optimization process for the three points are shown in Table 6.

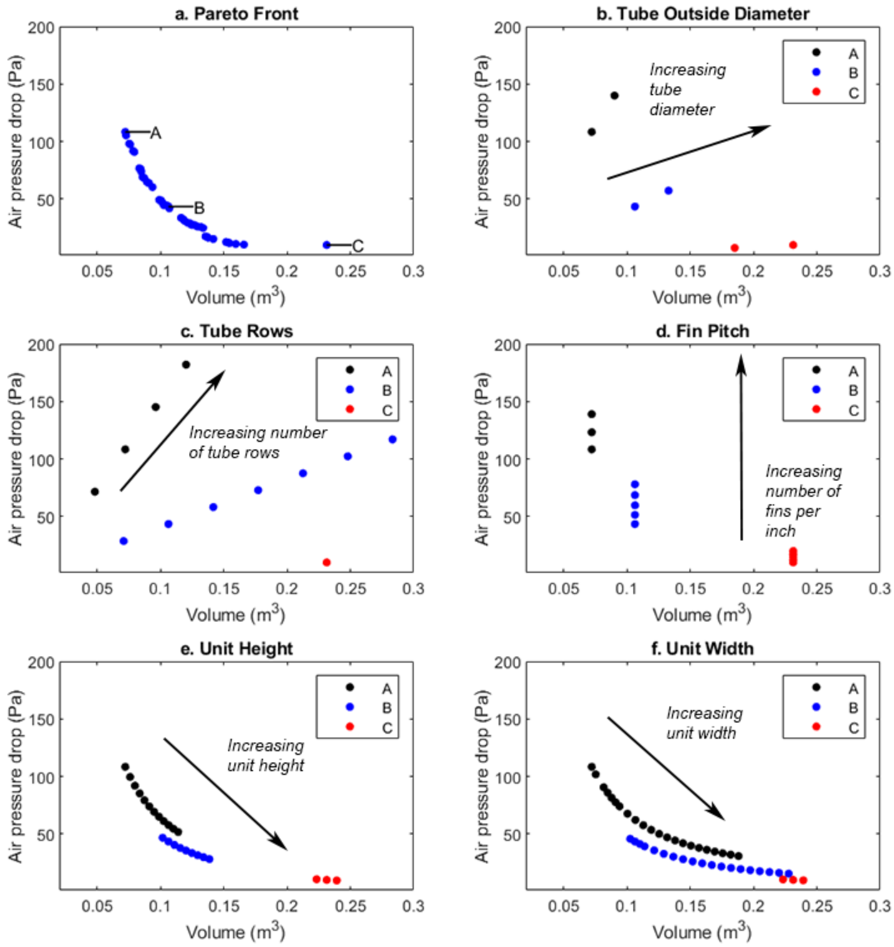

Figure 5 The variation of two objectives from three optimal Pareto solutions.

From Figure 5, the following optimization results can be obtained, such as:

a. Tube Outside Diameter

Figure 5b shows the influence of tube outer diameter on the air side pressure drop and the heat exchanger volume. It can be observed that the size of tube outside diameter proportionally affects the volume of the HE, whereas its effect towards the air side pressure drop gets weaker as the volume gets bigger. Point A shows steeper slope than point C.

b. Tube Rows

Figure 4c shows the influence of tube rows on the air side pressure drop and the HE volume. In all three cases of point A, B, and C, it can be observed that as the heat exchanger has higher number of rows, its air side pressure drop also increases. It can be concluded that the reason for this phenomenon is that higher rows means longer path for the air flow. Higher number of rows also increases the heat exchanger volume, because it means higher value for the heat exchanger unit length L.

c. Fin Pitch

Figure 4d shows the influence of the fin pitch on the air side pressure drop and the HE volume. In all three cases of point A, B, and C, it can be observed that as the fin pitch does not affect the heat exchanger value. However, higher number of fins per inch shows higher values of the air pressure drop. One reason for this is that denser fins means lesser air free flow area.

d. Unit Height and Weight

Figure 4e and figure 4f shows the influence of unit height and weight on the air side pressure drop and the HE volume. In all three cases A, B, and C, it can be observed that as the increasing value of unit height and weight is followed by the increasing value of the volume of the HE and the decreasing value of air side pressure drop. This may be caused by the larger frontal area for the air to pass through, therefore lesser air velocity and lesser air pressure drop.

Table 6 Three Pareto Optimal Solutions.

Parameters	A	B	C
Outer diameter (mm)	10.30	10.30	13.30
Number of rows	3.00	3.00	2.00
Fin pitch (fpi)	11.00	9.00	9.00
Unit width (m)	1.15	1.40	2.90
Unit height (m)	0.95	1.15	1.45
Overall heat transfer coeff. ($W/m^2.K$)	221	187	88
Air side heat transfer coeff. ($W/m^2.K$)	235.29	198.63	91.14
Water side heat transfer coeff. ($W/m^2.K$)	3642	3253	3004
Air flow velocity (m/s)	4.94	3.36	1.28
Water flow velocity (m/s)	1.38	1.13	1.00
Air pressure drop (Pa)	108.30	43.33	9.98
Water pressure drop (Pa)	2790	2188	2319
Total heat transfer surface area (m^2)	57.5	70.3	148.7
Annualized cost (IDR)	190,016,881	92,708,831	45,556,673
Relative error (%)	0.21	3.62	3.60

The Pareto front give a set of several optimum solutions, and it is up to the designer to choose one optimum solution for the optimized design. After comparing three optimum values (point A, B, and C), point B will be chosen because it fulfils all heat exchanger requirements and it has moderate air pressure drop compared to point A (lower annualized cost and fan noise) and it is way much smaller than point C, because the width of the unit represented by point B is almost half the width of the unit represented by point C. The comparison between the preliminary design using traditional approach and the optimized design using genetic algorithm is shown in Table .

Table 7 Design Parameter Comparison.

Parameters	Preliminary Design	Optimized Design
Outer diameter (mm)	13.3	10.3
Number of rows	3	3
Fin pitch (fpi)	12	9
Unit width **(m)**	1.1	1.4
Unit height **(m)**	0.8	1.15
Unit volume **(m^3)**	0.08	1.06
Overall heat transfer coefficient **($W/m^2.K$)**	206.17	187.15
Air flow velocity **(m/s)**	6.14	3.36
Water flow velocity **(m/s)**	1.23	1.13
Air side pressure drop **(Pa)**	222.12	43.33
Water side pressure drop **(Pa)**	9623	13436
Total heat transfer surface area **(m^2)**	61.56	70.28
Annualized cost **(IDR)**	304,528,545	92,708,831

The preliminary design was supposed to be an initial starting point, therefore, the constraints mentioned previously were not considered as it would be too complicated to fulfill multiple constraints at once with no prior heat exchanger designing experience. The values for the water flow velocity and water side pressure drop are acceptable; however, the airflow velocity of the preliminary design is way above the typical airflow velocity of 3 to 4 m/s, and the air side pressure drop does not meet the requirement of ASHRAE 61.2 standard. Higher air pressure drop will increase fan power consumption, which leads to a higher running cost of the heat exchanger and more fan noise. We can see that the estimated annualized cost of the preliminary design is relatively more expensive than the optimized design with a significant difference. Even though the unit size of the optimized design is bigger than the preliminary design, the optimized design fulfills all the constraints mentioned before. The annualized cost of the optimized design is only 30.4% of the preliminary design, and the air pressure drop can be reduced to 19.5% of the preliminary design, with only a 12.4% increase in volume. The final technical drawing of the optimum air cooler design is presented in Figure 6.

While these improvements are substantial, further research could focus on enhancing material efficiency and exploring the integration of renewable energy sources for even greater efficiency and sustainability. Additionally, the transferability of these optimized design parameters to other turbine models, such as those different from the OPRA OP-16, is promising. However, this would require adjustments to accommodate the specific operational characteristics of each turbine model. Future studies could explore these adaptations and validate their effectiveness across different turbine systems.

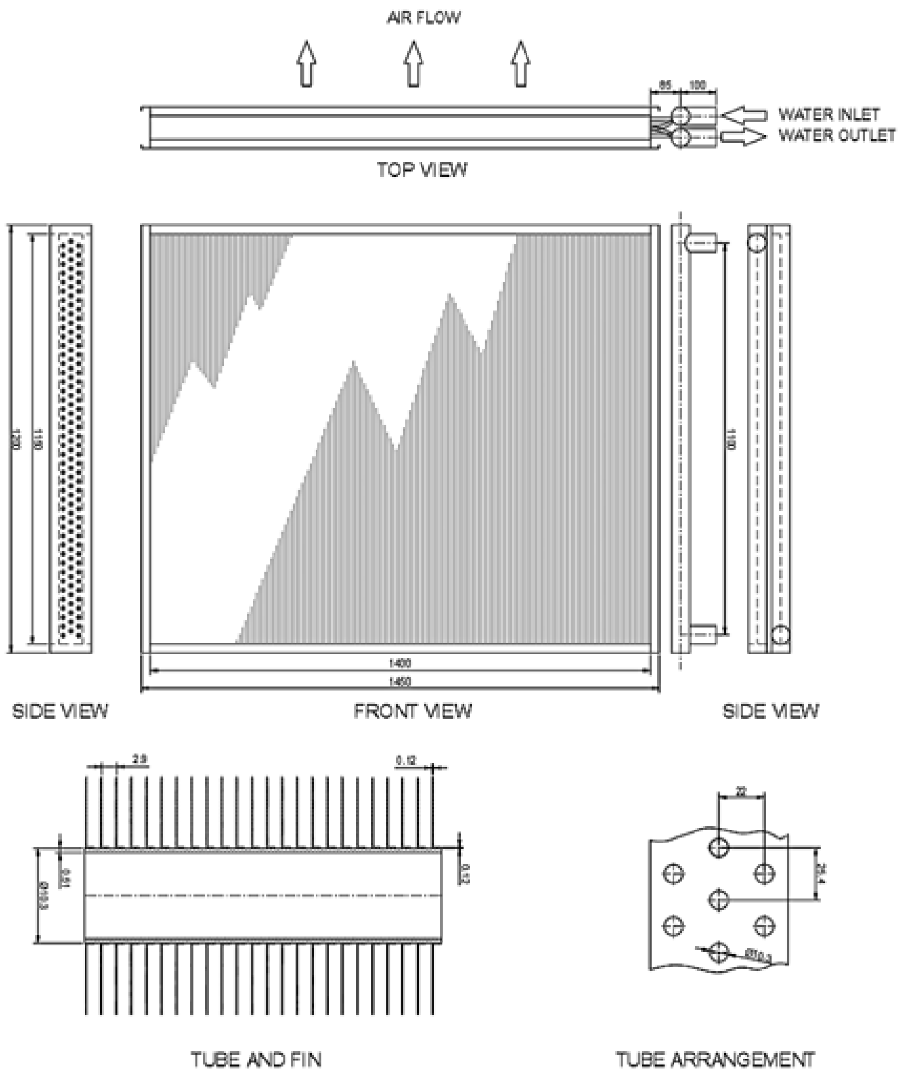

Figure 6 Technical drawing of the optimum air cooler design.

4. CONCLUSION

A mathematical modeling for finned-tube heat exchanger thermal design was built and a preliminary design was conducted using the traditional approach.

The optimization was conducted by implementing Non-Dominated Sorting Genetic Algorithm II (NSGA-II) with two objective functions: minimizing the heat exchanger volume and minimizing the air side pressure drop. The decision variables for the design were tube outer diameter, number of tube rows, fin pitch, unit height, and unit width.

A set of Pareto optimal points was obtained, and the Pareto front showed the trade-off between the two objective functions. The number of tube rows, unit width, and unit height were found to be important design parameters that caused the trade-off between the air pressure drop and heat exchanger volume. The other fin pitch only had a significant effect on the air pressure drop.

The calculated parameters and estimated cost of both preliminary design and optimized design were also compared. The preliminary design was undersized, resulting in a very high air pressure drop and, thus, a higher running cost. The optimized design offered a bigger alternative design while meeting all the constraints according to standards and industrial needs. The optimization reduced annualized cost to 30.4% and lowered air pressure drop to 19.5% with bigger heat exchanger volume of 12% compared to the preliminary design.

5. *Nomenclature*

A_o	=	Total heat transfer surface area (m^2)
$A_{air\ flow}$	=	Minimum free flow area (m^2)
D_o	=	Tube outer diameter (mm)
f_a	=	Friction factor of air
G_a	=	Mass velocity of air (kg.m/s)
H	=	Height of cooler unit (m)
L	=	Length of cooler unit (m)
W	=	Width of cooler unit (m)
$\rho_{a,i}$	=	Inlet air density (kg/m^3)
$\rho_{a,m}$	=	Mean air density (kg/m^3)
$\rho_{a,o}$	=	Outlet air density (kg/m^3)
σ	=	Ratio of free flow area and frontalarea

6. *References*

[1] Kyas, W.M. & London, A.L., *Compact Heat Exchangers, 3rd Edition*, Scientific International, 1984.

[2] OPRA Turbines, *OP16 Gas Turbine Genset Technical Leaflet*, 2020 (https://www.opraturbines.com/wp-content/uploads/2020/07/OPRA_Technical_Leaflet_English.pdf).

[3] Rao, R.V., Saroj, A., Ocloń, P. & Taler, J., *Design optimization of heat exchangers with advanced optimization techniques: a review*, Archives of Computational Methods in Engineering, **27**, pp. 517–548, 2020.
[4] Dastmalchi, M., Sheikhzadeh, G.A. & Arefmanesh, A., *Optimization of microfinned tubes in double pipe heat exchangers using particle swarm algorithm*, Applied Thermal Engineering, **119**, pp. 1-9, 2017.
[5] Lara-Montañoa, O.D. & Gómez-Castroa F.I., *Optimization of a shell-and-tube heat exchanger using the grey wolf algorithm*, Computer Aided Chemical Engineering, **46**, pp. 572-576, 2019.
[6] Rao, R.V. & Saroj, A., *Constrained economic optimization of shell-and-tube heat exchangers using elitist-Jaya algorithm*, Energy, **128**, pp. 785-800, 2017.
[7] Zareaa, H., Kashkoolic, F.M., Soltanic, M. & Rezaeianc, M., *A novel single and multi-objective optimization approach based on Bees Algorithm Hybrid with Particle Swarm Optimization (BAHPSO): Application to thermal-economic design of plate fin heat exchangers*, International Journal of Thermal Sciences, **129**, pp. 552–564, 2018.
[8] Mohanty, D.K., *Gravitational search algorithm for economic optimization design of a shell and tube heat exchanger*, Applied Thermal Engineering, **107**, pp. 184-193, 2016.
[9] Suyatna, N.D., "*Multi objective finned tube heat exchanger optimization using generic algorithm*", Under graduate thesis, Mechanical Engineering Depertment, Faculty of Mechanical and Aerospace Engineering, ITB, Bandung, 2022.
[10] Sanaye. S. & Hajabdollahi, H., *Multi-Objective Optimization of Shell and Tube Heat Exchangers*, Applied Thermal Engineering, **30**, pp.1937–1945, 2010.
[11] Fettaka, S., Thibault, J. & Gupta,Y., *Design of Shell-and-Tube Heat Exchangers Using Multiobjective Optimization*, International Journal of Heat and Mass Transfer, **60**(1), pp. 343–354, 2013.
[12] OPRA Turbines, *OP16 Gas Turbine Genset Technical Leaflet*, (Online), 2020 (https://www.opraturbines.com/wp-content/uploads/2020/07/OPRA_Technical_Leaflet_English.pdf).
[13] Hardi, D., *Project Pemasangan Gas Turbine Generator Opra OP16 Kapasitas 4 x 1,8 MW*, 2018. (https://dhevilsmechanic.blogspot.com/2018/09/pemasangan-gas-turbine-generator-4-x.html).
[14] DBM Coils, *Cooling Coils for Turbine*.
[15] Arora, R.K., *Optimization: Algorithms and Applications*, 2015.
[16] Kalyanmoy, D., *Multi-Objective Optimization Using Evolutionary Algorithms*, John Wiley & Sons, 2001.
[17] Ciconkov, R., *Refrigeration Software: Fin Coil Heat Exchanger*, Skopje, Macedonia, 2008. (Http://www.Unet.Com.Mk/Refrigeration/Index.Html)